

The effects of a multistep intercooled compression process implemented on a solar-driven Braysson heat engine



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ABSTRACT

The present study develops the thermodynamic analysis for the cycle of a solar-driven, Braysson cycle based plant in the ideal limit and in the presence of process irreversibilities. The plant cycle differs from the conventional idealized Braysson cycle in that the implementation of the final isothermal compression process is substituted by a multistep intercooled compression. The cycle's efficiency is analytically formulated after taking into account several loss (irreversibility) sources such as the non-isentropic behavior of the main compressor, the power turbine and the intercooled compressor stages as well as the actual heat transferred through countercurrent heat exchangers. All pressure losses associated with heat exchangers are related to the actual heat transfer load within each exchanger. The analysis develops a parametric evaluation for the effectiveness of the main cycle free variables on the thermal efficiency of the cycle. Such free variables include the working fluid maximum temperature, the compressor pressure ratio and the operating temperature limits of the intercooled compression stages, in addition to the polytropic coefficients of the compressor and power turbine (quasi-) isentropic processes. The results indicate that such a plant may reach efficiency levels above 30%, i.e. exceeding the efficiencies of the conventional Photovoltaic plants by a wide margin.

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1. Introduction

The enormous heat reservoir provided by the sun irradiation on the surface of the earth has been identified by many researchers and scholars as an energy source capable of supporting the operation of various Renewable and Clean Energy power plants. However, it was only during the last two to three decades that the technologies concerning solar power generation have attained maturity. Several research groups have investigated the coupling of a single or an array of solar collectors to heat engines (i.e. heat plants implementing a thermodynamic cycle). Numerous thermodynamic cycle models appropriate for these couplings were analyzed by such groups, adopting several different approaches. Within this framework, solar driven Carnot [1–5], solar driven Ericsson [6–9], solar driven Stirling [7,8,10–16] as well as solar driven Brayton [17–20] cycle plants have been investigated. These studies, mostly theoretical in nature, analyze via thermodynamic analysis, the impact of various irreversibilities upon the cycle performance and investigate the optimum criteria for the most energy efficient configuration.

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The Braysson cycle proposed by Frost et al. [21] is one of the most promising thermodynamic cycles that may easily be employed in solar power plants. As a straightforward extension to the gas turbine cycle, it is based on the combination of the best features of the conventional Brayton and Ericsson cycles. These are the high temperature isobaric heat addition provided by the Brayton, and the low temperature isothermal heat rejection adopted by the Ericsson cycle. The cycle is characterized by two processes. The first involves an extended expansion far below the atmospheric pressure, up to the point where the temperature of the fluid (i.e. the flue gases created inside the upstream combustion chamber) becomes nearly equal to that of the surrounding atmosphere. The second process (an isothermal re-compression of the cycle fluid back to the atmospheric pressure) allows for the discharge of these gases to the atmosphere. The major drawback for the cycle arises from the last process (i.e. the isothermal compression). So far, no mechanism has been proposed for a straightforward implementation of the process. However, a few such proposals were publicized by a number of researchers. One of them incorporates a regulated water injection, coordinated with the compression, so that the evaporation of the water may maintain the temperature of the fluid nearly constant [22].

The Braysson cycle constitutes essentially an extension to the well known Brayton cycle (upon which the operation of the gas

Nomenclature

T_i	temperature at corresponding station i (K)	L_I	length of the intercooler tubes (m)
P_i	pressure at corresponding station i (Pa)	$\bar{h}$	mean convective heat transfer coefficient (W/m ² K)
T_o	ambient temperature (K)	S_{tl}	Stanton number ($\bar{h}/\rho_l u_l C_p$)
T_A	inlet to the intercoolers fluid temperature (K)	C_{fl}	skin friction coefficient for intercooler tubes
T_B	exit of the intercoolers fluid temperature (K)	C_{FB}	skin friction coefficient for solar collector tubes
P_{C1}	pressure at the inlet of the stage intercoolers (Pa)	ρ_l	mean gas density over the exchanger (kg/m ³)
P_{C2}	pressure at the exit of the stage intercoolers (Pa)	u_l	mean bulk velocity of the cycle gas in stage intercoolers (m/s)
P_{C3}	pressure at the exit of the stage compressor (Pa)	F_c	concentration factor of the solar collectors
P_{ec}	pressure at the inlet of the exhausting leg of the pre-heater (Pa)	A_T	total area of the solar collectors (m ²)
P_o	ambient pressure = P_{at} (Pa)	q_s	specific area heat intensity of the solar illumination (W/m ²)
N_s	number of intercooled compression stages	n_B	solar collector heat absorption efficiency
N_c	number of compressor for multistage regenerative intercooled Ericsson	L_B	length of the tube running the solar collector (m)
N_T	number of turbines for multistage regenerative intercooled Ericsson	D_B	hydraulic diameter of the tube running the solar collector (m)
Π_{cs}	pressure ratio for the stage compression process	u_B	mean bulk velocity of the cycle gas in solar heater tubes (m/s)
W_{C1}	work transfer due to the compressor isentropic compression	ρ_B	mean gas density in solar heater tubes (kg/m ³)
W_e	work transfer due to the turbine isentropic expansion	P_{ec}	pressure at the exit of the final stage compressor (Pa)
$W_{int,comp}$	work transfer due to isentropic compression for stage compression	$W_{isothermal}$	work for isothermal process
$\dot{m}_a$	working fluid mass flux (kg/s)		
C_p	specific heat capacity (J/kg K) ($=C_{pc} = C_{pt}$)	Greek	
Q_{in}	heat transfer provided by the solar collectors (W)	Π_c	compressor pressure ratio
$\eta_{th,ID}$	ideal thermal efficiency of the cycle	$\Pi_s^{N_s}$	total intercooled pressure ratio over the intercooled compression stages
η_{th}	the real thermal efficiency of the plant	Θ	non-dimensional temperature ratio (T_3/T_o)
$\eta_{ID,Brs}$	ideal thermal efficiency of the Braysson cycle	Θ_A	non-dimensional temperature ratio (T_A/T_o)
$\eta_{ID,Brt}$	ideal thermal efficiency of the Brayton cycle	Θ_B	non-dimensional temperature ratio (T_B/T_o)
$\eta_{ID,Eri}$	ideal thermal efficiency of the Ericsson cycle	ΔP_I	pressure losses inside intercooler tubes
$\eta_{th,Ems}$	ideal thermal efficiency of the multistage regenerative intercooled Ericsson	ΔP_B	pressure losses inside solar heater tubes
$\eta_{th,Brs}$	the real thermal efficiency of the conventional Braysson cycle	$\Delta \Pi_B$	relative pressure loss term for the solar collector tubes ($\Delta P_B/P_2$)
e_{pt}	polytropic coefficient for power turbine	$\Delta \Pi_{ec}$	pressure losses inside the tubes of the exhausting leg of the preheater
e_{pc}	polytropic coefficient for the compressor	$\Delta \Pi_{C12}$	relative pressure loss term due to the heat transfer – cooling process ($\Delta P_I/P_{C1}$)
Q_{SI}	heat transfer in a single intercooling stage (W)		
D_I	hydraulic diameter of the intercooler tubes (m)		

turbines is based) and proposes a single gas equivalent for the Brayton–Rankine (air–steam) “Combined cycle” that dominates the design of the power plants that dominated the current market. The extension of the expansion processes removes the need for the heat exchanger that transports energy from the topping (i.e. the Brayton) to the bottoming cycle (i.e. the Rankine), replacing it by a bigger turbine. In other words, it eliminates the irreversibilities of the boilers and their ancillaries needed for the operation of the steam turbine/condenser plant. Although the Brayton cycle is among the most well studied cycles for air driven solar power stations, the reported studies on solar driven Braysson cycle plants are quite limited. So far, only the general performance characteristics of Braysson heat engines were investigated [23–27]. Due to the apparent absence of a reliable (easily implemented) isothermal compressor, these studies incorporate ideal isothermal compression processes, without any emphasis on the performance of the mechanism that supports its implementation. Of these studies, the most “solar” oriented may be considered the study by Zheng et al. [28], where a novel model of the solar-driven thermodynamic cycle system consisting of a solar collector and a Braysson heat engine is presented. Along similar lines, Wu and Lin [29] investigated the optimal performance and design parameters of an irreversible solar-driven Braysson heat engine, while Wu et al. [30] presented an irreversible solar-driven Braysson heat engine system

in which the temperature-dependent heat capacity of the working fluid, the radiation–convection heat losses of the solar collector and the irreversibilities resulting from heat transfer and non-isentropic compression and expansion processes were taken into account. A thermoeconomic based approach has been employed by Tyaki et al. [31] in order to investigate the performance of an irreversible Braysson heat engine affected by the finite-rate heat transfer between the working fluid and the heat reservoirs, heat leak loss from the heat source to the ambient and the irreversibility within the cycle. The analysis presented in the latter is general and can be useful for understanding the relationships and differences between the Braysson cycle and other cycles with respect to both, the economics and thermodynamics when employing different set of operating parameters and given constraints. Zhang and Lin [32] established a unified cycle model for a class of solar-driven heat engines in which a unified approach is introduced so that the cycle of the heat engine is composed of two adiabatic and two polytropic processes and may include the Brayton, Braysson, Carnot cycles, etc. Thus the results obtained are also general and consequently can be directly used to determine the optimal performance of any of the aforementioned cycles.

Recently Georgiou et al. [33] introduced the thermodynamic analysis of an ideal solar-driven near-Braysson cycle power plant concept, in which a feasible solution is provided for the isothermal

compression requirement. This is implemented by incorporating a multistage compression process which approaches a near isothermal state through intermediate intercooling among the stages. The analysis focused on the ideal cycle limit of the cycle processes and showed that the plant's optimum thermal efficiency may be found within the range of $45\% > \eta_{th, ID} > 60\%$, when the maximum temperature of the cycle is of order 1000 K, while the main free parameter being the maximum temperature that the solar collector and power plant materials may sustain. The intercooled compression process is not a completely new idea, since it was proposed (and analyzed) recently for increasing the performance of the Brayton cycle based Solar Plants [i.e. 34–36]. For solar based applications, it was discussed by Sanchez-Ortiz et al. [37] involving a cycle nearly equivalent to the Ericsson cycle, incorporating a sequence of Brayton cycles and based on similar studies made earlier by the same group [38,39]. In these works, the authors present the thermodynamic model and optimization of a sequence of multistep irreversible Brayton cycles, where the main objective was to propose a general analytical model-formalism, incorporating simultaneously the regeneration, multiple compressors and turbines as well as all the feasible irreversibility sources.

The current study focuses on the real cycle analysis for the solar driven, Braysson cycle based plant, incorporating a multistage intercooled compressor. This cycle provides a greater thermal efficiency and specific work when compared against the Brayton cycle, while it is much simpler than the conventional Brayton–Rankine combined cycle plants in its construction since it removes the boiler structure that is an absolute requirement for the latter plants. Against the Ericsson cycle it requires a much simpler structure, since it avoids the incorporation of a reheated multi stage turbine whose stages must operate at quite large temperatures. The Braysson cycle needs a simpler, continuous operation static heat exchanger structure on the high temperature end for its operation (when the Ericsson cycle splits it among the reheated expansion stages) and removes the large regenerative heat exchanger required for the operation of an Ericsson cycle plant. In addition the proposed cycle may be incorporated in large power output plants structures, a well known drawback of the Stirling cycle plants, since the available cycle implementation mechanism of the latter (i.e. the Ross Yoke mechanism) is quite difficult to be built in large sizes.

The analysis is carried out in terms of non-dimensional variables which provide a platform for quick estimates of the effectiveness of free variable deviations from a given “reference” configuration. The main compressor, power turbine and the intercooled multistage compressor stages are modeled as non-ideal (i.e. irreversible), while additional loss sources of the cycle are estimated analytically. These include the solar heater loss coefficient as well as the pressure losses due to the heat transfer – cooling process for the stage intercoolers. Thermal efficiencies above 30% are predicted for a real plant, i.e. nearly double that for the conventional photovoltaic (PV) plants.

2. The multistage intercooled, solar-driven near-Braysson cycle concept

The schematic configuration for the proposed plant is illustrated in Fig. 1. The power generating section of the plant is comprised of a number (N_s) of intercooled compression stages, attached to a common shaft along with the main compressor and the power turbine stages. The heat cycle is driven by means of the Solar radiation heat flux impinging on an array of solar thermal collectors units, which drive a heat exchanger that increases (at a nearly constant pressure) the temperature of the already compressed cycle gas (i.e. atmospheric air) up to the maximum level (T_3) permitted by the exchanger's materials. This high temperature fluid is then expanded within

the main turbine till it reaches a temperature (T_4) nearly equal to that of the surrounding atmosphere, generating along the way power through the electrical generator coupled to the same shaft. Since T_4 has been reduced to levels nearly equal to the atmospheric temperature, the pressure at the completion of the expansion (i.e. P_4) is well below the atmospheric. This very low pressure gas has to be re-compressed to a pressure somewhat above the atmospheric one, if this air is to be discharged back to the atmosphere. The discharge is implemented by passing the re-compressed gas through a countercurrent heat exchanger that preheats the atmospheric air entering the plant, so that the temperature of the latter rises from the atmospheric one to $T_A = T_4$. This process generates pressure losses for both air flows within the preheater and it is for this reason that the exhausting gas has to be re-compressed to a pressure somewhat above the atmospheric one, before entering the preheater. The re-compression process in the ideal Braysson cycle takes the form of an isothermal compression. Here it is proposed to implement the isothermal compression process by employing a multistage intercooled compressor operating within a small temperature band ($T_B < T < T_A$), where T_B is somewhat above the atmospheric (T_0), while T_A is equal to the starting temperature of the cycle. Each intercooler stage receives air at a temperature $T_4 = T_A$ and passes it through an isobaric cooling process (driven by the outside atmospheric air) down to the temperature T_B , which of course is somewhat greater than the ambient temperature T_0 . Each compression stage is completed by a quasi-isentropic compression of the air, raising the temperature from T_B back to T_A . A succession of intercooled compression stages is assembled, so that all stages are operating between the same temperature limits. If T_A is not too far above T_B the compression process does not differ too much from an isothermal one. The number of the required stages (N_s) is to be determined by the cycle's operating characteristics. In order to maintain the “isothermal” nature of the air during the re-compression, the air entering the main compressor (at station 1) must be preheated from the ambient temperature (i.e. T_0), up to the condition where $T_1 = T_A$ through the countercurrent regenerative preheater, driven by the exhaust from the last stage of the intercooled compressor.

3. Thermodynamic analysis of the ideal cycle

The ideal thermodynamic cycle of the proposed concept is depicted in Fig. 2 by means of a T - s diagram. The cycle is defined by the points 1-2-3-4 on this diagram and is assembled by the following processes:

- Process 1-2: isentropic compression
- Process 2-3: isobaric heating
- Process 3-4: isentropic expansion

The cycle incorporates N_s intercooled, compression stages where each stage implements two processes the intercooling followed by the isentropic compression. For example in the first stage the corresponding processes are

- Process 4-C: isobaric cooling
- Process C-D: isentropic compression

Defining P_i and T_i as the pressure and temperature respectively for each cycle station i (as defined in Fig. 2), in the ideal cycle limit the following equation relates the pressures between the stations 1, 2 and 3:

$$P_3 = P_2 = \Pi_c P_1 \quad (1)$$

where Π_c is the main compressor pressure ratio, equal to P_2/P_1 . In addition, it is assumed that

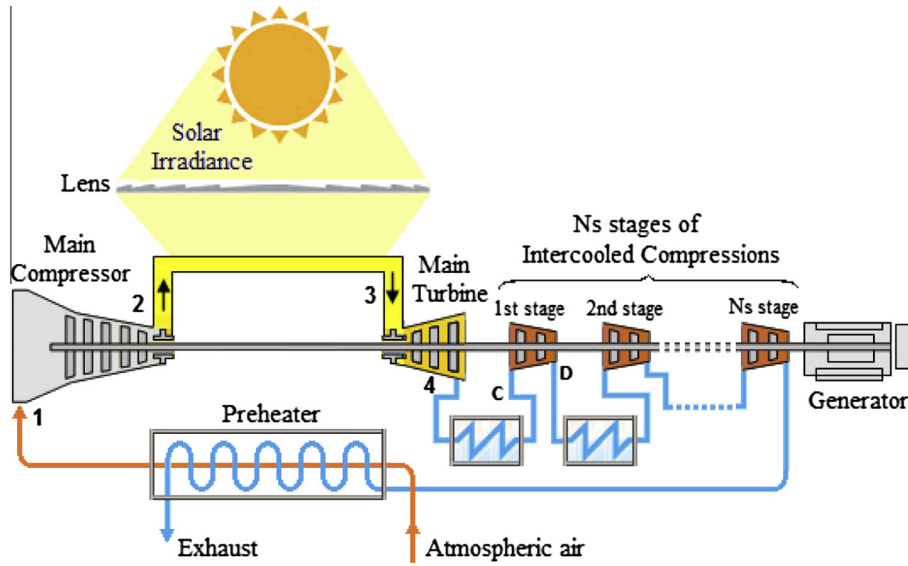


Fig. 1. Schematic configuration of the proposed concept.

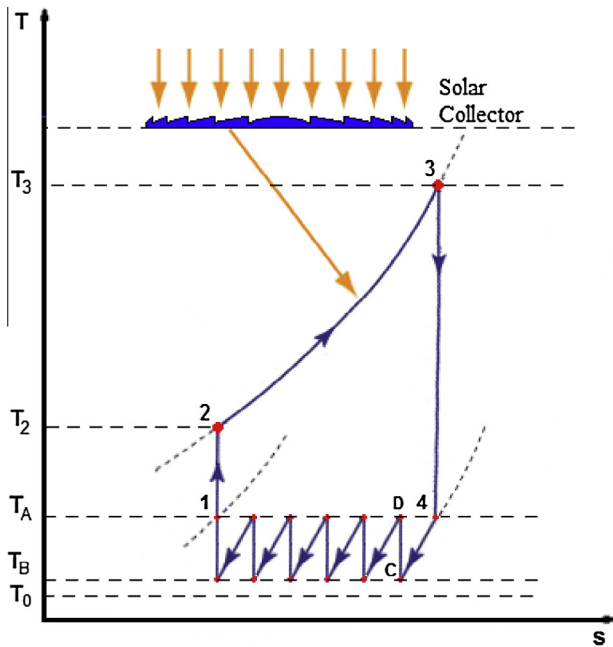


Fig. 2. T-s diagram of the ideal cycle.

$$T_4 = T_A = T_1 \quad (2)$$

The analysis is carried out in terms of the non-dimensional temperature parameters Θ_A , Θ_B , and Θ . These are derived by dividing the dimensional quantities by the reference temperature of the surrounding environment T_0 such as:

$$\Theta = \frac{T_3}{T_0} \quad (3)$$

$$\Theta_A = \frac{T_A}{T_0} \quad (4)$$

$$\Theta_B = \frac{T_B}{T_0} \quad (5)$$

The temperature at station 2 is given by:

$$T_2 = T_1 \Pi_c^{\frac{\gamma-1}{\gamma}} = T_1 \vartheta \quad (6)$$

where $\vartheta = \Pi_c^{(\gamma-1)/\gamma}$. Similarly, the pressure at station 4 may easily be deduced by:

$$P_4 = P_3 \left(\frac{T_4}{T_3} \right)^{\frac{\gamma}{\gamma-1}} = \Pi_c \left(\frac{T_4}{\Theta T_0} \right)^{\frac{\gamma}{\gamma-1}} P_1 = \Pi_c \left(\frac{(T_4/T_0)}{\Theta} \right)^{\frac{\gamma}{\gamma-1}} P_1 \quad (7)$$

In the multistage intercooled compression it is assumed that the number of stages is equal to N_s , each operating between the temperature limits T_A and T_B as indicated in Fig. 2. The pressure ratio (Π_{cs}) for the each stage is given by:

$$\Pi_{cs} = \frac{P_D}{P_C} = \frac{P_D}{P_4} = \left(\frac{T_A}{T_B} \right)^{\frac{\gamma}{\gamma-1}} \quad (8)$$

where P_D is the exit pressure of the stage compressor and it is derived through the isentropic process relationship as indicated below:

$$P_D = P_C \left(\frac{T_A}{T_B} \right)^{\frac{\gamma}{\gamma-1}} \quad (9)$$

Since temperatures T_A and T_B remain constant, the stage pressure ratio remains constant, equal to Π_{cs} . The “total” pressure ratio ($\Pi_{cs}^{N_s}$) of the intercooled compressor (all N_s stages) is equal to

$$\Pi_{cs}^{N_s} = \frac{P_1}{P_4} \quad (10)$$

Solving this equation for the N_s term, it leads to the following relationship:

$$N_s = \frac{\ln(P_1/P_4)}{\ln(\Pi_{cs})} = \frac{\ln \left[\frac{\Theta}{T_A/T_0} \right]^{\frac{\gamma}{\gamma-1}}}{\ln [T_A/T_B]^{\frac{\gamma}{\gamma-1}}} = \frac{\ln(\Theta/\Theta_A)}{\ln(\Theta_A/\Theta_B)} \quad (11)$$

It is clear from the above Eq. (11) that the number N_s is not a free parameter but depends on the selection of the operating temperature levels for the intercooled compressor.

3.1. The work (mechanical power) transfers

The work (actually the corresponding mechanical power) transfers W_{C1} and W_e due to the isentropic compression process 1–2 and isentropic expansion 3–4 respectively are given by:

$$W_{C1} = \dot{m}_a C_p (T_2 - T_1) = \dot{m}_a C_p T_0 \left(\Pi_c^{\frac{\gamma-1}{\gamma}} - 1 \right) \Theta_A \quad (12)$$

$$W_e = \dot{m}_a C_p (T_3 - T_4) = \dot{m}_a C_p T_0 (\Theta - \Theta_A) \quad (13)$$

In addition, the total work (power) transfer contributing to the isentropic compression in the various stages of the intercooled compressor is given by:

$$W_{C2} = N_s \dot{m}_a C_p (T_A - T_B) = \dot{m}_a C_p T_0 \frac{\ln(\Theta/\Theta_A)}{\ln(\Theta_A/\Theta_B)} (\Theta_A - \Theta_B) \quad (14)$$

3.2. The heat (power) input process

The heat (thermal power) input to the gas during the cycle process (2–3) (i.e. Q_{in}) as implemented at the solar collectors is given by:

$$Q_{in} = \dot{m}_a C_p (T_3 - T_2) = \dot{m}_a C_p T_0 \left(\Theta - \Theta_A \Pi_c^{\frac{\gamma-1}{\gamma}} \right) \quad (15)$$

3.3. The ideal thermal efficiency

Taking into account the above work (power) transfers and heat input, and after substituting for the energy transfer variables, the ideal thermal efficiency of the cycle ($n_{th,ID}$) is given by:

$$\begin{aligned} n_{th,ID} &= \frac{W_{net}}{Q_{in}} = \frac{W_e - (W_{C1} + W_{C2})}{Q_{in}} \\ &= 1 - N_s \frac{\ln(\Theta/\Theta_A)}{(\Theta/\Theta_A) - \Pi_c^{\frac{\gamma-1}{\gamma}}} \frac{1 - (\Theta_B/\Theta_A)}{\ln(\Theta_A/\Theta_B)} \end{aligned} \quad (16)$$

Considering the T - s diagram illustrated in Fig. 3a, one may identify the ideal cycles of Ericsson (e.g. 1-6-7-8-1), Brayton (e.g. 1-2-3-4-1) and conventional Braysson implementing a fully isothermal heat rejection process (e.g. 1-2-3-5-1). The ideal thermal efficiency of the corresponding conventional Braysson cycle $n_{ID,Br}$ can be easily deduced as:

$$n_{ID,Brs.} = 1 - \frac{Q_{OUT}}{Q_{IN}} = \frac{W_{exp} - (W_{C1} + W_{C2})}{Q_{IN}} \quad (17)$$

where W_e is the work transfer during the expansion process 3–5 and W_{C1} and W_{C2} the corresponding work transfers due to the compression process 1–2 and heat output process 5–1 respectively, as it is illustrated in Fig. 3a. Thus, after executing the necessary calculations, the efficiency can be written as:

$$n_{ID,Brs.} = 1 - \left(\frac{\gamma}{\gamma-1} \right) \frac{\ln \left(\frac{P_1}{P_5} \right)}{\Theta - \Pi_c^{\frac{\gamma-1}{\gamma}}} \quad (18)$$

Setting as $\Theta = \Pi_c^{\frac{\gamma-1}{\gamma}}$, then the efficiency finally becomes:

$$n_{ID,Brs.} = 1 - \frac{\ln \left(\frac{\Theta}{\Theta} \right)}{(\Theta - \Theta)} \quad (19)$$

Working similarly for the Ericsson and Brayton cycles the corresponding ideal thermal efficiencies can be written in a similar form as:

$$n_{ID,Eri.} = 1 - \frac{1}{\Theta} \quad (20)$$

$$n_{ID,Brt.} = 1 - \frac{1}{\Pi_c^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{\Theta} \quad (21)$$

In addition, for an ideal multistage regenerative intercooled Ericsson cycle (Fig. 3b), the ratio of N_c compressors to N_T turbines needed to run the plant is easily deduced as

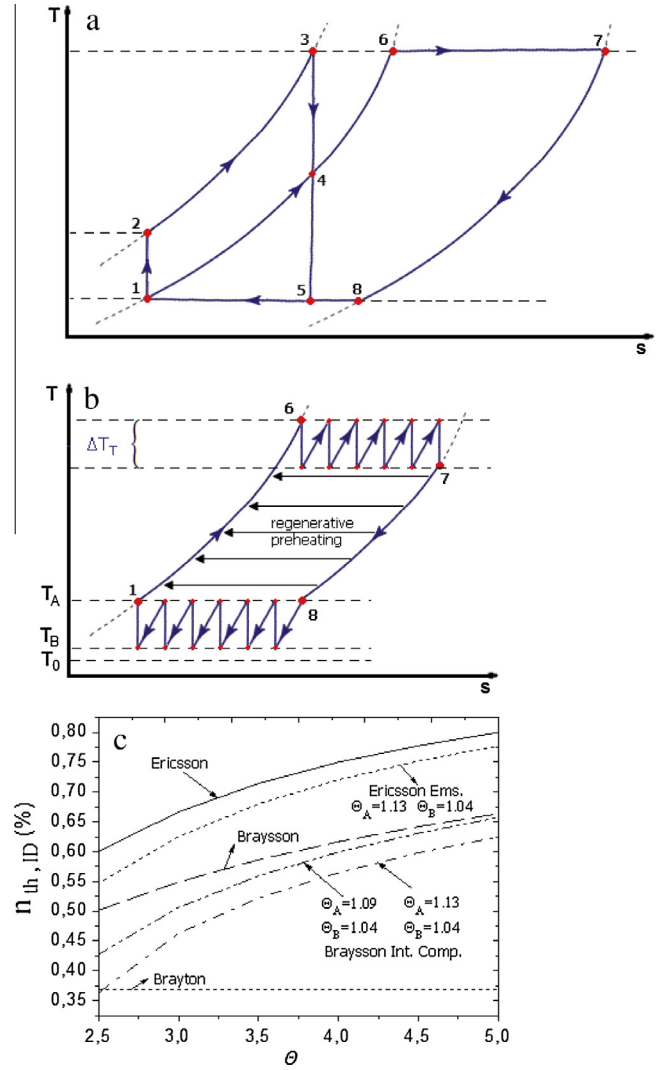


Fig. 3. (a) Ideal thermal efficiency of the proposed cycle as a function of Θ , in comparison to Ericsson, Braysson, Brayton and regenerative intercooled Ericsson efficiencies for the same pressure ratio $\Pi_c = 5$, (b) the T - s diagram of the ideal cycles, and (c) the T - s of the ideal regenerative intercooled Ericsson.

$$\frac{N_c}{N_T} = \frac{\ln \left\{ \frac{1}{1 - (\Delta\Theta_T/\Theta)} \right\}}{\ln(\Theta_A/\Theta_B)} \quad (22)$$

where $\Delta\Theta_T$ is the non-dimensional term of temperature drop ΔT_T during the expansion process in the turbines, i.e. $\Delta\Theta_T = \Delta T_T/T_0$. The corresponding efficiency can be written as a function of the ratio N_c/N_T as

$$n_{ID,Ems} = 1 - \left(\frac{N_c}{N_T} \right) \left(\frac{\Theta_A - \Theta_B}{\Delta\Theta_T} \right) \quad (23)$$

Fig. 3c illustrates the ideal thermal efficiency $n_{th,ID}$ of the proposed cycle as a function of Θ . This efficiency is compared to the ideal thermal efficiency of the Ericsson ($n_{ID,Eri.}$), the Brayton ($n_{ID,Brt.}$), the conventional Braysson ($n_{ID,Brs.}$) and the multistage regenerative intercooled Ericsson cycle ($n_{ID,Ems}$) operating within the same temperature limits (i.e. Θ) and pressure ratio Π_c . A more comprehensive comparison on the performance of the ideal Braysson cycle against those of the Ericsson and the Brayton cycles is presented in Appendix A of the present paper.

The data illustrated in Fig. 3c show that the ideal thermal efficiency of the multiple intercooled compression Braysson cycle

$n_{th,ID}$ (i.e. the one investigated in the current study), lies between the efficiency levels of the conventional (i.e. fully isothermal compression) Braysson ($n_{ID,Brt.}$) and the conventional Brayton ($n_{ID,Brt.}$) cycles. The multistage intercooled compression seems to approximate well the perfectly isothermal one. When temperature T_A is restricted to lower values (e.g. $\Theta_A = 1.09$) the efficiency levels reach even closer to that of the conventional Braysson. Of course the efficiency of the Ericsson cycle lies well above of the other cycles, although the efficiency here was calculated on the basis of ideal isothermal expansion–compression processes. If these were to be substituted by multi isentropic expansion/isobaric reheat and isentropic compression/isobaric cooling stages, the efficiency of the Ericsson cycle would have been quite smaller than the one presented here.

4. The real cycle analysis

The T - s diagram of the real cycle is illustrated in Fig. 4. By assuming again that $T_4 = T_A$ and $P_2 = \Pi_C P_1$, the pressures at stations 1 and 2 are given by the following relationships:

$$P_1 = (1 - \Delta\Pi_I)P_o \quad (24)$$

$$P_3 = (1 - \Delta\Pi_B)P_2 \quad (25)$$

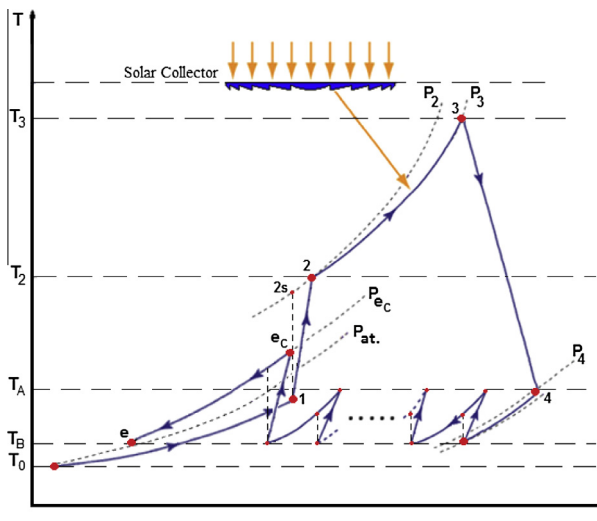
Here $\Delta\Pi_I$ and $\Delta\Pi_B$ are the relevant pressure loss coefficients for the flows inside the tubes within the inlet leg of the preheater and the solar collector. On the other hand, the losses inside the tubes of the exhausting leg of the preheater demands an additional over-compression by the intercooled compressor (defined by the coefficient $\Delta\Pi_{ec}$), so that the re-compressed air may be exhausted into the atmosphere, i.e. the process $e_c - e$ in Fig. 4. In other words

$$P_{ec} = (1 + \Delta\Pi_{ec})P_1 \quad (26)$$

The pressure at station 4 is given by

$$P_4 = P_3 \left(\frac{T_4}{T_3} \right)^{\frac{\gamma-1}{\gamma-1e_{pt}}} \quad (27)$$

where e_{pt} represents the polytropic coefficient for the expansion process.



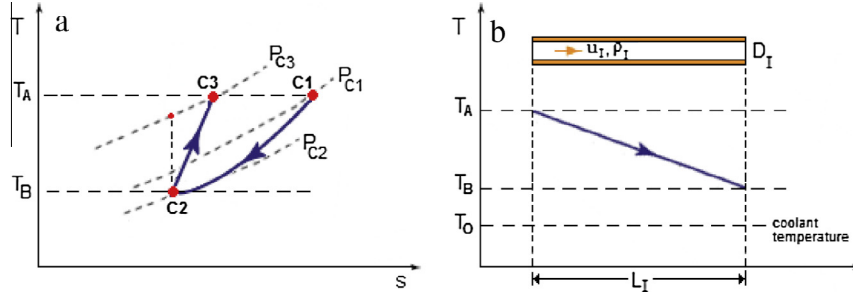


Fig. 5. (a) The single stage intercooled compression processes and (b) the air cooling process assuming $T_{coolant} = T_o = T_{innerwall}$ (i.e. water cooling).

Hence, the pressure loss term ΔP_I of Eq. (33) takes the following form:

$$\Delta P_I = \frac{1}{2} \rho_I u_I^2 \left(\frac{C_{fI}}{4S_{tI}} \right) \ln \left[\frac{\Theta_A - 1}{\Theta_B - 1} \right] \quad (36)$$

In addition, the relative pressure loss factor $\Delta \Pi_{C12}$ is now given by

$$\Delta \Pi_{C12} = \frac{\Delta P_I}{P_{C1}} = \frac{1}{8} \frac{u_I^2}{RT_A} \left(\frac{C_{fI}}{S_{tI}} \right) \ln \left[\frac{\Theta_A - 1}{\Theta_B - 1} \right] \quad (37)$$

In most heat exchanger applications the value of the skin friction coefficient C_{fI} over that of the Stanton number S_{tI} is usually found to be within the range $2 < C_{fI}/S_{tI} < 4$ ([40]). The velocity u_I may be adjusted among stages so as to maintain ΔP_I constant. Under such an assumption it may easily be deduced that the “total” intercooled compression ratio $\Pi_s^{N_s}$ for the sum of the intercooled compression stages may be expressed by:

$$\Pi_s^{N_s} = \frac{P_{ec}}{P_4} = (1 + \Delta \Pi_{ec}) \frac{P_1}{P_4} \quad (38)$$

Hence, an expression for the required stages number N_s may be established as follows:

$$N_s = \frac{\ln \left[(1 + \Delta \Pi_{ec}) \frac{P_1}{P_4} \right]}{\ln \left[(1 + \Delta \Pi_{C12}) \left(\frac{T_A}{T_B} \right)^{\frac{1}{\gamma-1} e_{pc}} \right]} \quad (39)$$

4.2. The solar heater relative pressure loss coefficient $\Delta \Pi_B$

Fig. 6 gives an illustration of the solar heater collector tube, operating between the limits of temperatures T_2 and T_3 . Based on the geometric characteristics of the tube and the gas flow inside, the pressure losses inside the tube are given by:

$$\Delta P_B = \frac{1}{2} \rho_B u_B^2 \left(\frac{L_B}{D_B} \right) C_{FB} = (1 - \Delta \Pi_{C12}) \left(\frac{T_A}{T_B} \right)^{\frac{1}{\gamma-1} e_{pc}} \quad (40)$$

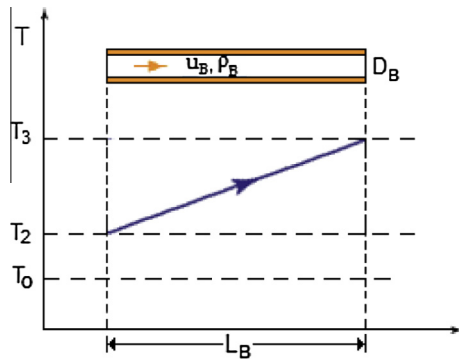


Fig. 6. The solar heater collector tube process.

where C_{FB} is the skin friction coefficient inside the tubes, while from the energy conservation equation one may deduce that

$$\dot{m} C_p (T_3 - T_2) = F_c L_B D_B n_B q_s \quad (41)$$

In Eq. (41) F_c represents the concentration factor of the solar collector defined by $F_c = A_T / L_B D_B$, where A_T is the total outer area of the collectors facing the sun while L_B , D_B represent the length and diameter of the tubes carrying the cycle fluid and positioned along the focal line of the collector. In addition q_s represents the specific area heat intensity of the solar illumination and n_B the solar heat absorption efficiency. By a simple rearrangement, Eq. (41) leads into:

$$\frac{L_B}{D_B} = \frac{\pi \rho_B u_B C_p (T_3 - T_2)}{4 F_c n_B q_s} \quad (42)$$

Hence

$$\Delta P_B = \frac{\pi}{8} \rho_B^2 u_B^3 \frac{C_p (T_3 - T_2)}{F_c n_B q_s} C_{FB} \quad (43)$$

and

$$\Delta \Pi_B \cong \frac{\Delta P_B}{P_2} = \frac{\pi}{8} \frac{\rho_B^2 u_B^3}{RT_2} \frac{C_p (T_3 - T_2)}{F_c n_B q_s} C_{FB} \quad (44)$$

4.3. The real cycle work (mechanical power) transfers

The corresponding work (power) transfers for the real cycle are defined as

$$W_{C1} = \dot{m} C_{pC} T_{at} \left(\Pi_c^{\frac{\gamma_c-1}{\gamma_c} \frac{1}{e_{pc}}} - 1 \right) \quad (45)$$

for the main compressor

$$W_{ex} = \dot{m} C_{pT} T_{at} (\Theta - \Theta_1) \quad (46)$$

for the main turbine expansion and

$$W_{int.comp.} = N_s \dot{m} C_{pT} T_{at} (\Theta - \Theta_1) \quad (47)$$

for the intercooled compression stages. Here C_{pC} and C_{pT} are the specific heats for the compressions and turbine expansion process respectively and are considered equal and constant at C_p throughout the cycle.

4.4. The real plant efficiency

Finally the efficiency of the entire real plant can now be easily defined by

$$n_{th} = \frac{W_{net}}{q_{in}} = \frac{(\Theta - \Theta_1) - \Theta_1 \left(\frac{C_{pC}}{C_{pT}} \right) \left[\Pi_c^{\frac{\gamma_c-1}{\gamma_c} \frac{1}{e_{pc}}} - 1 \right] - N_s (\Theta_1 - \Theta_o)}{\left[\Theta - \Theta_1 \Pi_c^{\frac{\gamma_c-1}{\gamma_c} \frac{1}{e_{pc}}} \right]} \quad (48)$$

The corresponding real efficiency for the conventional Braysson cycle incorporating an isothermal recompression of the flue gases during the heat rejection process may be easily derived:

$$n_{th,Brs} = \frac{W_{net}}{Q_{in}} = 1 - \frac{\ln \left[\frac{\frac{\Theta}{\theta^{e_{pc} e_{pt}}}}{[1 - \Delta T_B]^{\frac{\gamma-1}{\gamma} e_{pt}}} \right]}{[\Theta - \theta]} \quad (49)$$

where in this case $\theta = \Pi_c^{\frac{\gamma-1}{\gamma} e_{pc}}$

5. Results and discussion

In order to restrict the size of the present study, two different case scenarios were investigated with respect to the expected range of the cycle operating characteristics, i.e.: the standard (st) one (where the parameters are assigned values well within the range available by current turbomachinery component technology) and the maximum (mx) one (where the parameters are assigned the best values within reach for the current component technology), as shown in Table 1. In the first case scenario (st), typical values of e_{pc} and e_{pt} were employed for the compressor and turbine polytropic coefficients respectively, which corresponds to conventional values for compressors and power turbines used in current technology gas turbines [41]. For the second case scenario (i.e. mx), a more efficient pair of compressor and turbine was considered, employing higher polytropic coefficients, as those reported by recent technology reviews [42]. The ambient temperature T_o was set at 303 K and 288 K respectively for the two case scenarios (the first corresponding to a hot day, while the second corresponds to an average day temperature), while the temperature difference between the fluid exiting the intercoolers and the ambient temperature ($T_B - T_o$), was maintained constant at 10 K for both scenarios by properly altering the respective temperature values. In addition, for both cases, the maximum temperature T_3 of the cycle is restricted within temperatures that may be sustained by conventional High Temperature steels. In other words it was assigned two values i.e. 900 K (For conventional steels) and 1050 K for Nickel–Chrome alloys (i.e. $\Theta \cong 3$ to $\Theta \cong 3.6$).

The skin friction coefficient to the Stanton number ratio was set equal to $C_{f1}/S_{st} = 2$ which is typical for the majority of low pressure loss heat exchanger designs [40]. For the solar collectors it was assumed that the concentration factor F_c is equal to 20.0. This value corresponds to the majority of Single-Axis Line Concentrators summarized by Wyman et al. [43]. The magnitude of the specific area heat intensity of the solar illumination q_s was assumed to be equal to 1000 W/m^2 while the solar absorption efficiency was set as $n_B = 0.9$. For all processes of the working fluid, the specific heat capacity C_p was kept constant and equal to 1005 J/kg K , since Wu et al. [30] have shown that the effect of variable specific heat capacity with respect to the temperature is less than 2% on the efficiency of solar cycles. The same implies for the specific heat capacity of the “coolant” fluid which runs the stage intercoolers.

Fig. 7 illustrates the thermal efficiency n_{th} of the cycle as a function of the compressor's pressure ratio Π_c for both case scenarios. The fluid temperature T_A was fixed at 353 K so that the intercoolers operated with a temperature difference of 40 K for the standard

(st) case and 55 K for the maximum (mx) case. The data indicate that for the (mx) case, the cycle's thermal efficiency may be found within the 35% and 42% range (depending upon the maximum temperature T_3 selected). As expected, lower thermal efficiencies were obtained for the (st) case lying between 20% and 27%. An optimum pressure ratio Π_c value exists for obtaining the maximum efficiency, but this is not constant. Considering the scenarios illustrated in Fig. 7, the optimum Π_c lies between 2 and 4.

The corresponding efficiency $n_{th,Brs}$ of the conventional Braysson cycle when the component irreversibilities have been taken into account and assuming an isothermal re-compression for the heat rejection process is illustrated in Fig. 8 as a function of pressure ratio Π_c . In order for the efficiency to be comparable to that of the proposed cycle (Fig. 7) the maximum temperature T_3 employed is of the order of 900 K and 1050 K which again leads to $\Theta \cong 3.0$ and $\Theta \cong 3.6$. For the polytropic coefficients of power turbine and compressor, the values employed for the standard (st) and maximum (mx) cases were employed here too. The calculated efficiency levels are now maintained below the 50% threshold for both, the standard (st) and maximum (mx) case when compared to the ideal Braysson efficiency levels illustrated in Fig. 3c. Of course, as it is expected, adopting a lower value of Θ (i.e. $\Theta \cong 3.0$) results in lower efficiency levels.

The results illustrated in Fig. 9 imply that the influence of the temperature T_A is not very significant (within the range $333 \text{ K} < T_A < 353 \text{ K}$) when the compression ratio is near optimum (e.g. $\Pi_c = 3$)

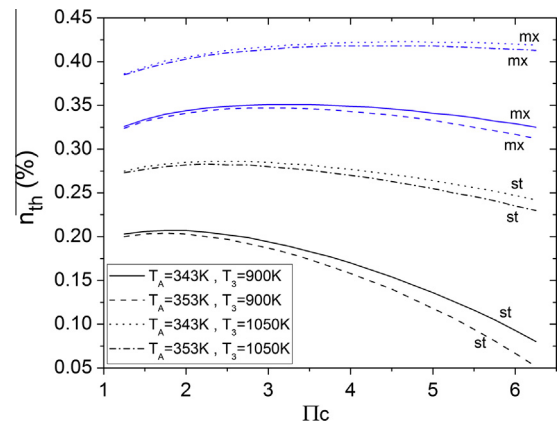


Fig. 7. The plant's thermal efficiency n_{th} as a function of pressure ratio Π_c for $T_A = 343 \text{ K}$ and $T_A = 353 \text{ K}$.

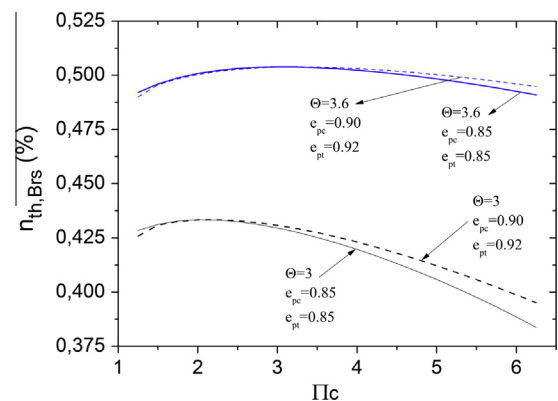


Fig. 8. The conventional Braysson thermal efficiency $n_{th,Brs}$ as a function of pressure ratio Π_c for different values of Θ .

Table 1
The characteristics of the two case scenarios investigated.

Parameter	Standard case (st)	Maximum case (mx)
T_3	900 K, 1050 K	900 K, 1050 K
e_{pc}	0.85	0.90
e_{pt}	0.85	0.92
T_B	313 K	298 K
T_o	303 K	288 K

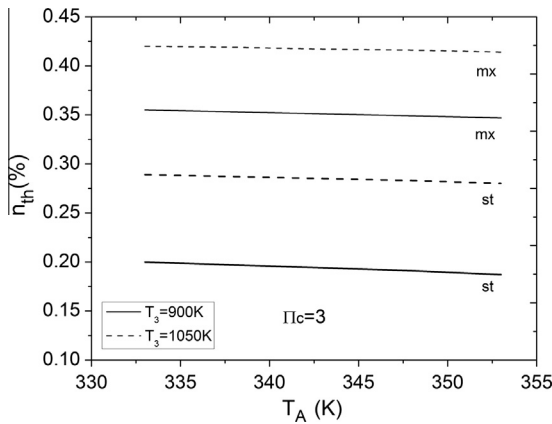


Fig. 9. The plant's thermal efficiency η_{th} as a function of temperature T_A for constant pressure ratio $\Pi_c = 3$.

The required number of intercooled stages in the (nearly) isothermal compressor (i.e. the parameter N_s) is exhibited by the data illustrated in Fig. 10, when N_s is plotted against the pressure ratio Π_c for $T_A = 343$ K and $T_A = 353$ K. It is quite clear that the mx scenario requires fewer stages and that a similar tendency is observed for the rising main compressor pressure ratio. In addition, it is very obvious that greater polytropic efficiencies in the turbomachinery components reduce the number of intercooled compression stages drastically. The influence of a greater T_A is toward a reduced N_s as T_A increases. In any case, the plant will need 5–8 stages to operate efficiently. Finally, as T_3 increases the number of N_s increases as well.

The magnitude of the gas pressure at the completion of the expansion (against the atmospheric pressure) is illustrated in Fig. 11, for the two scenarios and different Π_c and T_3 values. It is clear that the higher polytropic efficiency mx scenario leads to higher P_4 pressure levels (requiring a smaller intercooled compressor). The opposite is observed for the rising T_3 parameter. In any case, if the plant is to operate within the optimum compression ratio (i.e. maximum thermal efficiency) range ($3 < \Pi_c < 4$), the P_4/P_0 ratio is comparable to condenser pressures observed in current steam plants. The influence of the magnitude of the Π_c parameter is almost linear in increasing the P_4/P_0 ratio.

The free parameter influence on the heat input through the solar collector tubes is illustrated in Fig. 12. This input is evaluated through the temperature difference between the points T_2 and T_3 . This difference increases with T_3 and decreases with Π_c .

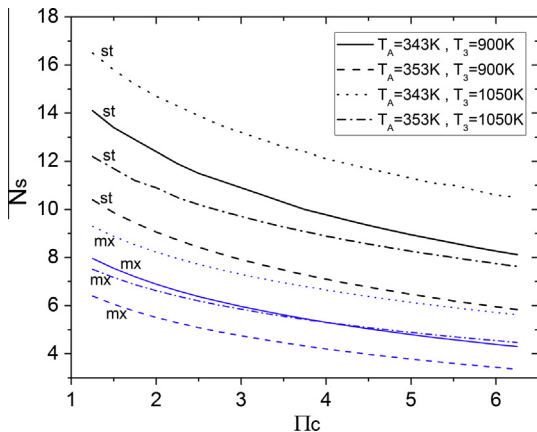


Fig. 10. The number of intercooled compression stages N_s required as a function of pressure ratio Π_c for $T_A = 343$ K and $T_A = 353$ K.

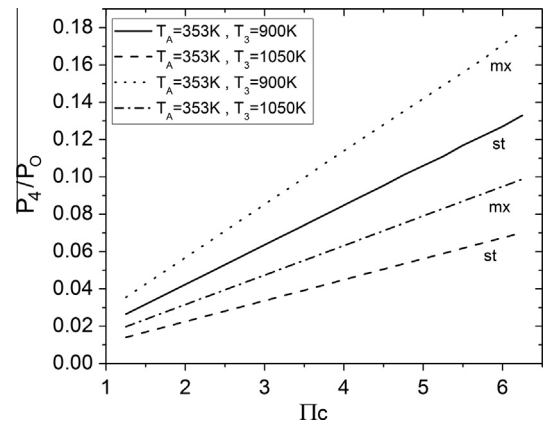


Fig. 11. The influence of pressure ratio Π_c on the pressure ratio P_4/P_0 for $T_A = 353$ K.

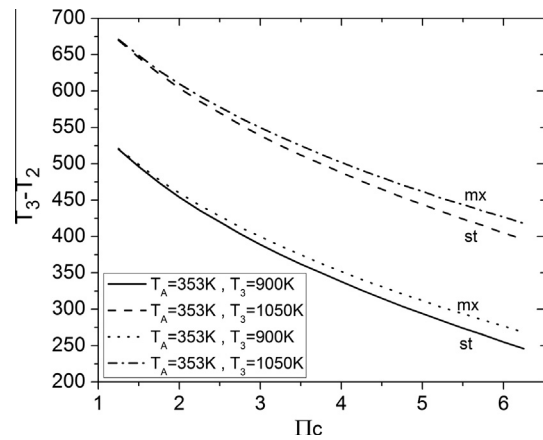


Fig. 12. The influence of pressure ratio Π_c on the temperature difference $T_3 - T_2$.

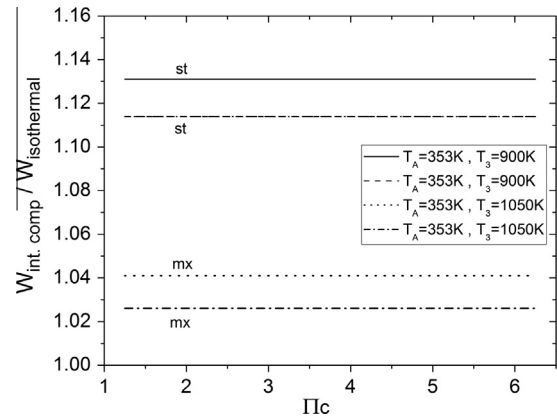


Fig. 13. The multistep compression to isothermal work ratio $W_{int.comp.}/W_{isothermal}$ as a function of pressure ratio Π_c .

The variation of the ratio $W_{int.comp.}/W_{isothermal}$ of the actual multistep intercooled compression work ($W_{int.comp.}$) over that of a corresponding perfectly isothermal process ($W_{isothermal}$) is shown in Fig. 13 as a function of Π_c which implies no significant influence on the ratio. In other words, for the (st) case the multistep compression increases the compression work by nearly 10–15% against the perfectly isothermal process and almost 2.5–4% for the (mx) case.

6. Conclusions

In the present study, a solar driven Braysson power cycle incorporating multistage intercooled compression stages is investigated via detailed thermodynamic analysis. The study provides an analysis of the real cycle as well as some aspects of the ideal cycle. Analytic expression is obtained for the thermal efficiency after determining all the cycle's irreversibility sources. The influence of the dominant free parameters are investigated upon the cycle's efficiency by applying two case scenarios regarding the operating conditions, such as the operating cycle temperatures, the pressure ratio, the operating limits of the intercooled compression stages and the characteristic polytropic coefficients of the compressor and power turbine. Results show that real thermal efficiency may exceed 40%, (i.e. for (mx) case) and it can be further extended if more "exotic" materials are employed for the mechanical components of the cycle that would be exposed at higher temperatures provided by the solar illumination.

The concept proposes a feasible solution for implementing the Braysson's cycle provision for an isothermal compression by the use of the multistage intercooled compression process.

Appendix A. Comparison on the performance of the ideal Braysson cycle against those of the Ericsson and the Brayton cycles

Based upon Fig. A1 the following ideal cycles may be distinguished:

- The Braysson (Brs) cycle: 1-2BR - 3BR - 5BR - 1
- The Ericsson (Eri) cycle: 1-2ER - 3ER - 4ER - 1
- The Brayton (Brt) cycle: 1-2BR - 3BR - 4BR - 1

The corresponding ideal thermal efficiency of each cycle is derived as

$$n_{ID,Brs.} = 1 - \frac{\ln(\frac{\theta}{\Theta})}{(\Theta - \theta)} \quad (A1)$$

for the Braysson cycle,

$$n_{ID,Eri.} = 1 - \frac{1}{\Theta} \quad (A2)$$

for the Ericsson cycle (where $\theta = T_{2BR}/T_1 = T_{max}/T_{min}$) and

$$n_{ID,Brt.} = 1 - \frac{1}{\Pi_c^{\gamma-1/\gamma}} = 1 - \frac{1}{\theta} \quad (A3)$$

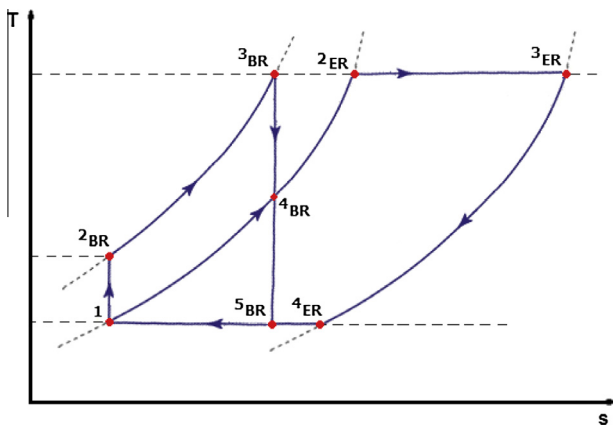


Fig. A1. T-s diagram of the ideal processes of Braysson, Brayton and Ericsson cycles.

for the Brayton cycle, where $\theta = T_{2BR}/T_1$ and $\Pi_c = P_{2BR}/P_1$. The heat (thermal power) input for each case can be written with respect to parameters θ and Θ as:

$$Q_{in,Brt} = \dot{m}_a C_p (T_{3BR} - T_{2BR}) = \dot{m}_a C_p T_1 (\Theta - \theta) \quad (A4)$$

$$Q_{in,Eri} = \dot{m}_a R T_{2ER} \ln \left(\frac{P_{2ER}}{P_{3ER}} \right) = \dot{m}_a C_p T_1 (\Theta - \theta) \quad (A5)$$

$$Q_{in,Brs} = \dot{m}_a C_p (T_{3BR} - T_{2BR}) = \dot{m}_a C_p T_1 (\Theta - \theta) = Q_{in,Brt} \quad (A6)$$

Now, by assuming that the three cycles receive the same amount of heat input it is implied that $Q_{in,Brs} = Q_{in,Brt} = Q_{in,Eri}$. After executing the necessary calculations and solving for the pressure ratio P_{2ER}/P_{3ER} , then the above condition is satisfied when:

$$\frac{P_{2ER}}{P_{3ER}} = e^{(1-\frac{\theta}{\Theta})\frac{\gamma}{\gamma-1}} \quad (A7)$$

The corresponding net work outputs for the three cycles under the assumption of the same heat input are given by:

$$W_{n,Brt.} = n_{ID,Brt} Q_{in,Brt} = \dot{m}_a C_p T_1 (\Theta - \theta) \left(1 - \frac{1}{\theta} \right) \quad (A8)$$

$$W_{n,Brs} = n_{ID,Brs} Q_{in,Brs} = \dot{m}_a C_p T_1 (\Theta - \theta) \left[1 - \frac{\ln(\frac{\theta}{\Theta})}{\Theta - \theta} \right] \quad (A9)$$

$$W_{n,Eri} = n_{ID,Eri} Q_{in,Eri} = \dot{m}_a C_p T_1 (\Theta - \theta) \left(1 - \frac{1}{\Theta} \right) \quad (A10)$$

where $W_{n,Brt.}$, $W_{n,Brs.}$, $W_{n,Eri}$ represent the net work outputs for the Brayton, Braysson and Ericsson cycles respectively. Hence, the following expressions may be extracted:

$$\frac{W_{n,Brs}}{W_{n,Brt}} = \frac{\theta}{(\Theta - 1)} \left[1 - \frac{\ln(\frac{\theta}{\Theta})}{\Theta - \theta} \right] \quad (A11)$$

$$\frac{W_{n,Brs}}{W_{n,Eri}} = \frac{\Theta}{(\Theta - 1)} \left[1 - \frac{\ln(\frac{\theta}{\Theta})}{\Theta - \theta} \right] \quad (A12)$$

Fig. A2 illustrates a comparison of the net work output for the Ericsson and Brayton ideal cycles by means of their respective work ratio against the net work output of the ideal Braysson cycle, i.e. $W_{n,Brs}/W_{n,Brt}$ and $W_{n,Brs}/W_{n,Eri}$. Focusing in Fig. A2 the following observations can be made:

- The net work output of the Braysson cycle is much larger (of the order of twice the magnitude) than the corresponding output of the Brayton cycle.

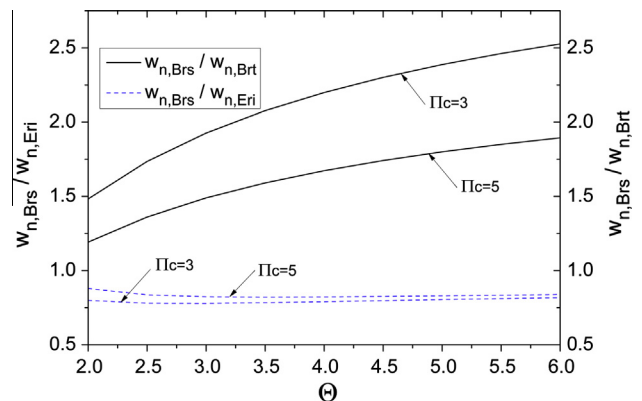


Fig. A2. A comparison of the net work output of the Ericsson, Brayton and Braysson cycles.

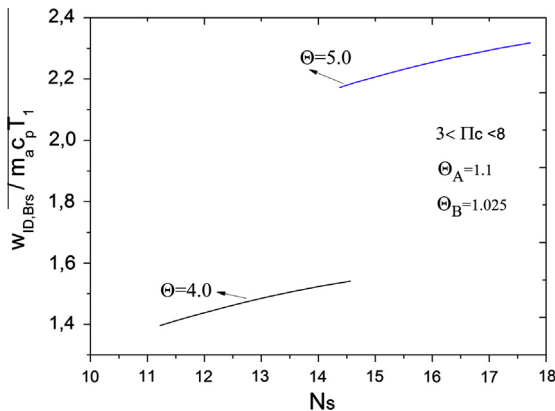


Fig. A.3. The comparison of the net work output of the Braysson cycle against the required number of intercooled compressor stages.

- (b) The net work output of the Braysson cycle is roughly equal to 85% the corresponding magnitude of the Ericsson cycle.
- (c) The increased net work output of the Braysson cycle decreases when the main compressor ratio increases.

In general, smaller main compressor ratios leads to increased thermal efficiency and net work output for the Braysson cycle, the opposite to what is observed for the Brayton one.

In addition, Fig. A.3 shows a comparison of the ideal Braysson net work output $W_{n,BrS,I.C}$ (incorporating the multiple intercooled compression stages, N_s) against the number of the required number of stages N_s . Here, the main variables are the parameters Θ and Π_c . The results indicate that:

- (a) The non-dimensional net work output ($W_{ID,BrS}/\dot{m}_a C_p T_1$) changes rather little (of the order of $\pm 5\%$ over the entire range of the main compressor pressure ratio Π_c) for the entire range of the number of stages N_s required for the plant operating conditions.
- (b) The range of N_s varies by a greater extent over the given range of compression ratios. This phenomenon increases significantly as the parameter Θ is increased from $\Theta = 4$ to $\Theta = 5$.

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